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HEAT TRANSFER DURING IN-TUBE CONDENSATION OF CFC-113 WITH DOWNFLOW IN VERTICAL, INTERNALLY-ENHANCED TUBES

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ABSTRACT

Accurate repeatable heat-transfer measurements have been made for condensation of CFC-113 with downflow inside enhanced "micro-fin" tubes. The heat-transfer rate was calculated from the coolant flow rate and temperature rise, the latter measured using a ten-junction thermopile with careful attention paid to adequate coolant mixing and isothermal immersion of the thermopile leads. The surface temperature was found from thermocouples embedded in the tube wall. One plain tube and nine enhanced tubes with different fin heights, helix angles and number of starts were tested. Enhancement ratios (i.e. vapour-side, heat-transfer coefficient for the enhanced tube divided by that for a smooth tube at the same vapour-side temperature difference and vapour inlet velocity) between 1.6 and 5.6 were found, with values depending on vapour-side temperature difference and vapour inlet velocity.

NOMENCLATURE

A	Constant in Eqn. (1)
h_{fg}	Specific enthalpy of evaporation
L	Length (height) of test tube
n	Constant in Eqn. (1)
q	Mean heat flux based on inside surface area of test tube, using fin root diameter for enhanced tube
Re	Condensate film Reynolds number ($4\Gamma/\mu$)
U_i	Vapour velocity at tube inlet
α	Vapour-side heat-transfer coefficient
ε	Enhancement ratio i.e. ratio of vapour-side heat-transfer coefficients for plain and enhanced tubes at same vapour inlet velocity and temperature difference
ΔT	Vapour-side temperature difference
Γ	Condensate mass flow rate per width at tube exit (qL/h_{fg})
μ	Viscosity of condensate

INTRODUCTION

In recent years a range of internally enhanced tubes, using low "micro-fins", has become available. Khanpara et al. (1986) reported heat-transfer data for condensation of CFC-113 inside eight micro-fin tubes. All tubes had between 60 and 70 fins per circumference with fin heights from 0.1 to 0.19 mm. Enhancement ratios between 1.9 and 2.3 were found, with higher fins generally giving higher enhancement ratios. Schlager et al. (1990) condensed HCFC-22 inside micro-fin tubes with fin heights up to 0.3 mm. Enhancement ratios between 1.5 and 1.8 were found. As before, larger fin heights gave higher enhancement ratios, with fin helix angle also having some effect. In a more recent study, Liu (1997) presented local heat-transfer data for one micro-fin tube condensing HFC-134a and HCFC-22.

In all of the above, the advantage of micro-fin tubes over plain tubes is evident. General conclusions, however, are difficult to draw due to the number of (particularly geometric) parameters involved. More data are needed in order to identify the important parameters and to develop general correlations. In this preliminary report, heat-transfer results are presented for nine enhanced tubes and a plain tube tested with condensation of CFC-113 in vertical downflow. In particular, fins heights up to 0.68 mm were tested, significantly higher than hitherto reported.

Special attention has been paid to experimental accuracy. Work is continuing on a wider range of experimental parameters, including tubes with inserts, condensation in the presence of a non-condensing gas, condensation of mixtures and pressure drop measurements.

EXPERIMENTAL METHOD

The stainless-steel and glass test apparatus, shown schematically in Fig. 1, consisted of a closed loop, with vapour generated in an electrically-heated boiler (maximum power 16 kW). The vapour was directed vertically downward through a calming section, before flowing through the test tube which was cooled externally by water flowing through an annulus in counterflow. Excess vapour passed to an auxiliary condenser from which the condensate returned to the boiler by gravity. CFC113 was used as the condensing fluid and all tests were done at a little above atmospheric pressure with three vapour velocities at entry (6.3 m/s, 7.5 m/s and 9.0 m/s).

The cooling water temperature rise, from which the heat-transfer rate to the test tube was calculated, was measured using a 10-junction thermopile. Care was taken to ensure adequate mixing and isothermal immersion of the thermopile leads in the vicinity of the junctions. A small predetermined correction for the dissipative temperature rise of the cooling water in the annulus and mixing boxes was incorporated in the calculation of the heat-transfer rate. The estimated accuracy of the measurement of the coolant temperature rise was better than 0.01 K. (All thermocouples were calibrated in a high precision constant temperature bath against a platinum resistance thermometer, accurate to 0.005 K. The accuracy of the thermo-emf measurement was 2 μV, equivalent to 0.005 K for the ten-junction thermopile.) The range of coolant temperature rise was 1.1 K to 3.2 K. The coolant flow rate was measured using a variable-aperture, float-type flow meters with an accuracy better than 2%. The vapour velocity at approach to the test section was found from the measured power input to the boiler, with a small, predetermined correction for the heat loss from the well-insulated boiler and supply pipe to the test section (see Lee and Rose, 1984).

The tubes tested in the present work were enhanced internally by low, spiral fins with trapezoidal profiles, supplied by . Nine different enhanced tubes were tested with different fin heights, fin pitches and helix angles. A plain tube was also tested for comparison. The tube dimensions are given in Table 1.

Both the plain and enhanced tubes were instrumented with three thermocouples embedded in the tube wall positioned 187.5, 375 and 562.5 mm from the vapour inlet (top of tube). The wall thermocouples were placed in 25 mm long slots in the outer tube wall and covered with soldered copper strips. The thermocouple leads were led out through the coolant via holes in the end bushes of the coolant annulus.

TABLE 1 - TEST TUBE DIMENSIONS

Tube No	Number of fins	Helix angle* / degree	Outside diameter / mm	Fin root diameter / mm	Fin tip diameter / mm	Fin height / mm	Wall thickness / mm	Circum-ferential fin pitch / mm	Fin tip half angle / degree
A1	36	25	25.0	23.0	21.6	0.68	1.0	2.01	30
A2	36	25	25.2	23.0	22.0	0.51	1.1	2.01	30
A3	36	25	25.3	23.0	22.3	0.36	1.2	2.01	30
B1	22	25	24.5	21.8	21.1	0.35	1.34	3.11	61
B2	22	25	24.4	21.8	20.9	0.43	1.31	3.11	61
B3	22	25	24.4	21.9	20.7	0.62	1.22	3.13	61
C1	54	45	24.9	22.1	21.7	0.22	1.40	1.29	25
C2	54	45	24.8	22.1	21.5	0.31	1.35	1.29	25
C3	54	45	24.7	22.1	21.2	0.44	1.31	1.29	25

* Measured from a line parallel to the tube axis

RESULTS AND DISCUSSION

All tubes were tested twice at each vapour inlet velocity to ensure repeatability. For tube A1 and the plain tube Figs. 2 and 3 show respectively, the dependence of heat flux and vapour-side heat-transfer coefficient, both based on the fin-root diameter, on vapour-side temperature difference. The data shown include tests on different days and show excellent repeatability; the enhancement due to the fins can clearly be seen. Similar results, but with smaller enhancements, were found for the other eight enhanced tubes. The range of vapour-side temperature difference for each tube varied somewhat due to different coolant inlet temperatures on different days.

When the data for the plain tube were compared to the models of Shekriladze and Gomelauri (1966) and Fujii and Uehara (1972) for laminar, forced-convection condensation on a vertical, flat plate, it was seen that the heat-transfer coefficients were up to 3 times higher than theory probably due to ripples and turbulence in the condensate film. The range of condensate Reynolds number ($4\Gamma/\mu$) for the plain tube data was 530 to 1300, well above the critical value for turbulent film flow in forced-convection condensation but below the critical value for turbulent film flow in free-convection (see, for instance Rohsenow and Choi, 1961). When compared to the free-convection, "laminar-wavy" film correlation of Kutetaladze (1963), the present heat-transfer coefficients were 2 to 3 times higher than the correlation. The data for inlet velocities of 6.3 and 7.5 m/s were roughly in line with the free-convection, turbulent film models of Labuntsov (1960) and Colburn (1934) respectively, while the data for inlet velocity 9 m/s were approximately 50% higher than the Colburn correlation. When equations of the form

$$q = A\,\Delta T^{n} \tag{1}$$

were fitted to the data it was found that for the plain tube n was in all cases close to 1, while for the enhanced tubes n was found to be close to 0.5. For tube A1 and the plain tube Figs. 2 and 3 show best-fit lines with n set to these values. Using equation (1) with n = 1 for the plain tube and n = 0.5 for the enhanced tubes we have

$$\varepsilon = \left(\frac{q_{enhanced}}{q_{plain}}\right)_{\text{same } \Delta T \text{and} U_i} = \left(\frac{\alpha_{enhanced}}{\alpha_{plain}}\right)_{\text{same } \Delta T \text{and} U_i} = \left(\frac{A_{enhanced}}{A_{plain}}\right)\Delta T^{-0.5} \quad (2)$$

where $A_{enhanced}$ and A_{plain} relate to the respective constants in equation (1). From Equation (2) it is seen that the enhancement ratio, as defined, is a function of vapour-side temperature difference and vapour inlet velocity. (The latter because $A_{enhanced}$ and A_{plain} depend on vapour inlet velocity.) Since in Fig. 2 the origin is a point on all curves, equation (1) for the plain tubes, with the corresponding values of A, should extrapolate satisfactorily to lower values of ΔT. The values of ε given by equation (2) should therefore be reliable for the range of ΔT used in the present tests. Table 2 lists the values of A for each tube and vapour inlet velocity together with the corresponding enhancement ratios for ΔT = 12 K (where the plain and enhanced tube data overlap) and ΔT = 4 K (the lower bound of the enhanced tube data). From Table 2 it can be seen that the best of the nine tubes was tube A1 which, for a vapour inlet velocity of 6.3 m/s, had an enhancement ratio of 5.6 at ΔT = 4 K and 3.2 at ΔT = 12 K.

Figure 4 shows the variation of enhancement ratio with fin height for all the tubes and for a vapour inlet velocity of 7.5 m/s. It may be noted that each point on Fig. 4 represents tests on two days and that if values of ε are found separately for each day, virtually identical results are obtained. A clear dependence on fin height can be seen with higher fins giving greater enhancement. The exception to this is found when comparing the two C tubes with lowest fin heights, which gave almost identical enhancement. The reason for this is not clear but it is possible that for fins less than about 0.3 mm high the interfin space may be completely flooded with condensate retained by surface tension, resulting in no improvement with fin height up to this value. Comparing the A and B tubes it can be seen that that more fins (36 for the A tubes, 22 for the B tubes) results in greater enhancement for a given fin height as expected. On this basis one would expect the C tubes, with 54 fins, to be significantly better than the A tubes for a given fin height but this is not the case and could be due to the higher helix angle on the C tubes. A small helix angle (i.e. fins running closer to the axial direction) will aid condensate drainage both by gravity in the present vertical tubes and by vapour shear.

From Table 2 it can be seen in most cases that enhancement ratio decreases with vapour velocity, indicating that vapour shear has a weaker effect on the enhanced tubes than on plain tube. The effect is strongest for the most highly enhanced tubes and weakest for the least enhanced tube.

CONCLUSIONS

A set of internally enhanced tubes has been carefully tested for condensation of downward flowing CFC-113. Enhancement ratios (defined as the heat flux or heat-transfer coefficient for the enhanced tube divided by that of the plain tube at the same vapour-side temperature difference and vapour inlet velocity) in the range 2.7 to 5.6 for ΔT = 4 K and 1.6 to 3.2 for ΔT = 12 K were found, with enhancement ratio decreasing with increasing vapour inlet velocity. The results also indicated that, for the present range of vapour-inlet velocity and vapour-side temperature difference, the enhancement ratio was inversely proportional to the square root of the vapour-side temperature difference. As noted earlier, this is a preliminary report of a much broader investigation. Further results will be reported in due course.

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TABLE 2 - SUMMARY OF RESULTS

Tube	Vapour velocity at inlet / (m/s)	A (see Eqn. (1))*	$\left(\frac{A_{enhanced}}{A_{plain}}\right) / K^{1/2}$	ε ($\Delta T = 4$ K, see Eqn. (2))	ε ($\Delta T = 12$ K, see Eqn. (2))
Plain	6.3	1.12	1.00	-	-
Plain	7.5	1.25	1.00	-	-
Plain	9.0	1.61	1.00	-	-
A1	6.3	12.52	11.18	5.59	3.23
A1	7.5	12.82	10.26	5.13	2.96
A1	9.0	14.05	8.73	4.36	2.52
A2	6.3	10.50	9.38	4.69	2.71
A2	7.5	11.00	8.80	4.40	2.54
A2	9.0	12.52	7.78	3.89	2.24
A3	6.3	8.60	7.68	3.84	2.22
A3	7.5	9.40	7.52	3.76	2.17
A3	9.0	10.60	6.58	3.29	1.90
B1	6.3	6.40	5.71	2.86	1.65
B1	7.5	7.31	5.85	2.92	1.69
B1	9.0	8.56	5.32	2.66	1.53
B2	6.3	6.92	6.18	3.09	1.78
B2	7.5	7.59	6.07	3.04	1.75
B2	9.0	8.82	5.48	2.74	1.58
B3	6.3	8.45	7.54	3.77	2.18
B3	7.5	8.90	7.12	3.56	2.06
B3	9.0	9.63	5.98	2.99	1.73
C1	6.3	7.81	6.97	3.49	2.01
C1	7.5	8.97	7.18	3.59	2.07
C1	9.0	10.37	6.44	3.22	1.86
C2	6.3	7.91	7.06	3.53	2.04
C2	7.5	8.87	7.18	3.59	2.07
C2	9.0	10.23	6.35	3.18	1.83
C3	6.3	10.38	9.27	4.63	2.68
C3	7.5	11.57	9.26	4.63	2.67
C3	9.0	12.78	7.94	3.97	2.29

* Note: For the plain tube A has units ($kW/m^2 K$) and for the enhanced tubes units ($kW/m^2 K^{1/2}$)

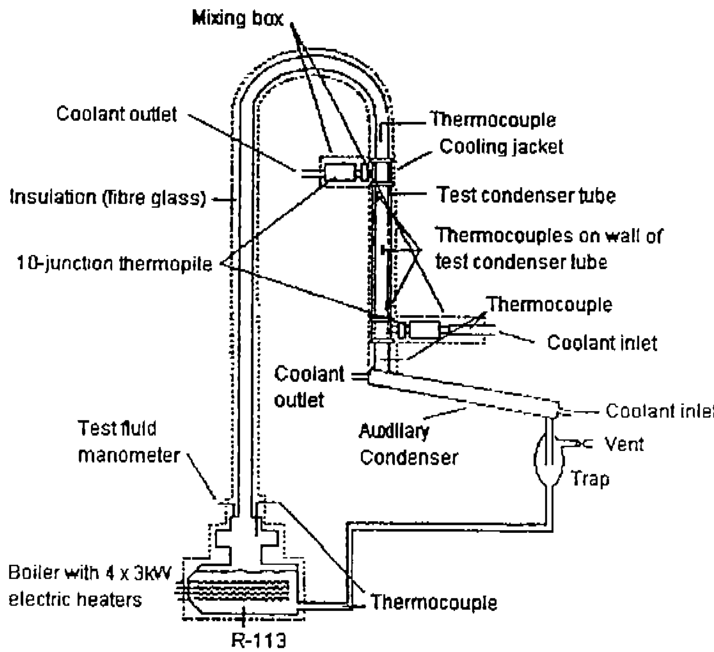

Figure 1 - Apparatus

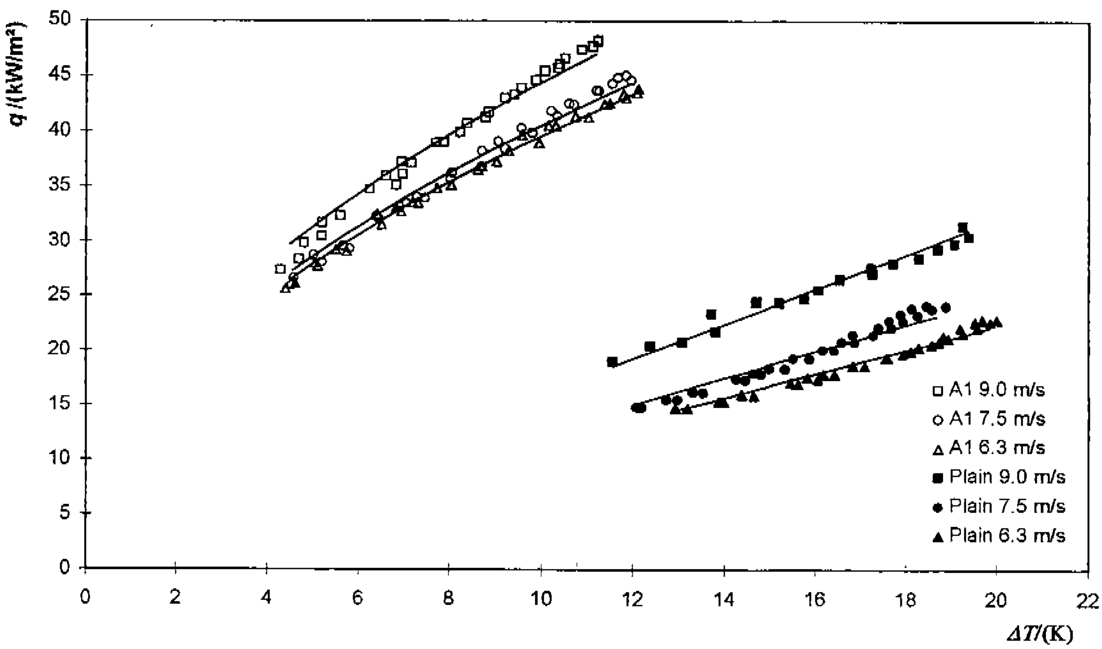

Figure 2 - Vapour-side results (q v ΔT) : Tube A1

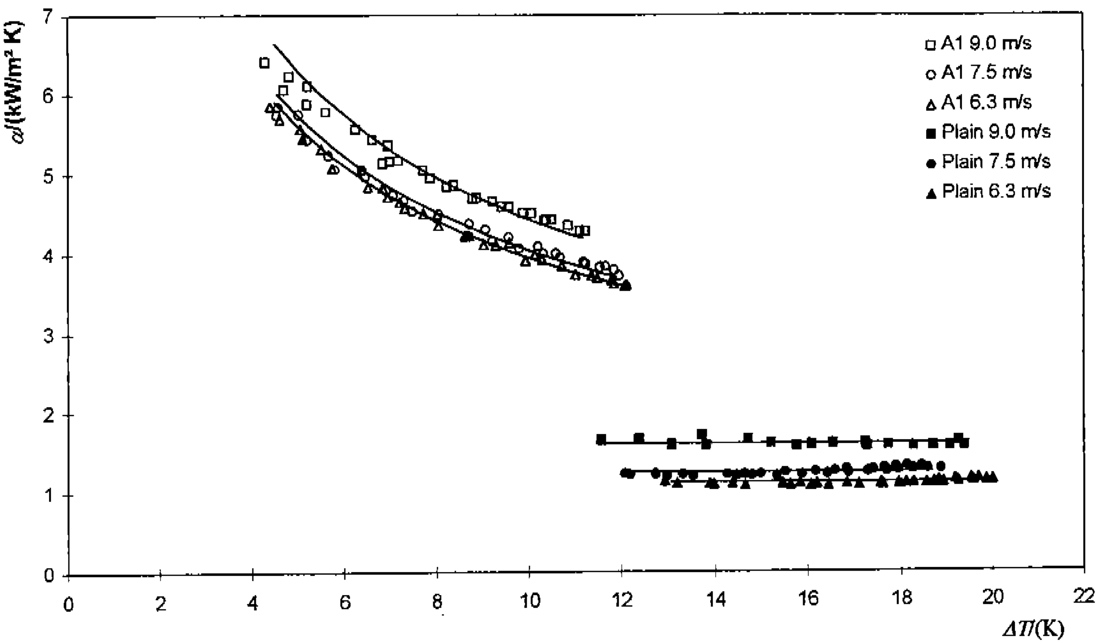

Figure 3 - Vapour-side results (α v ΔT) : Tube A1

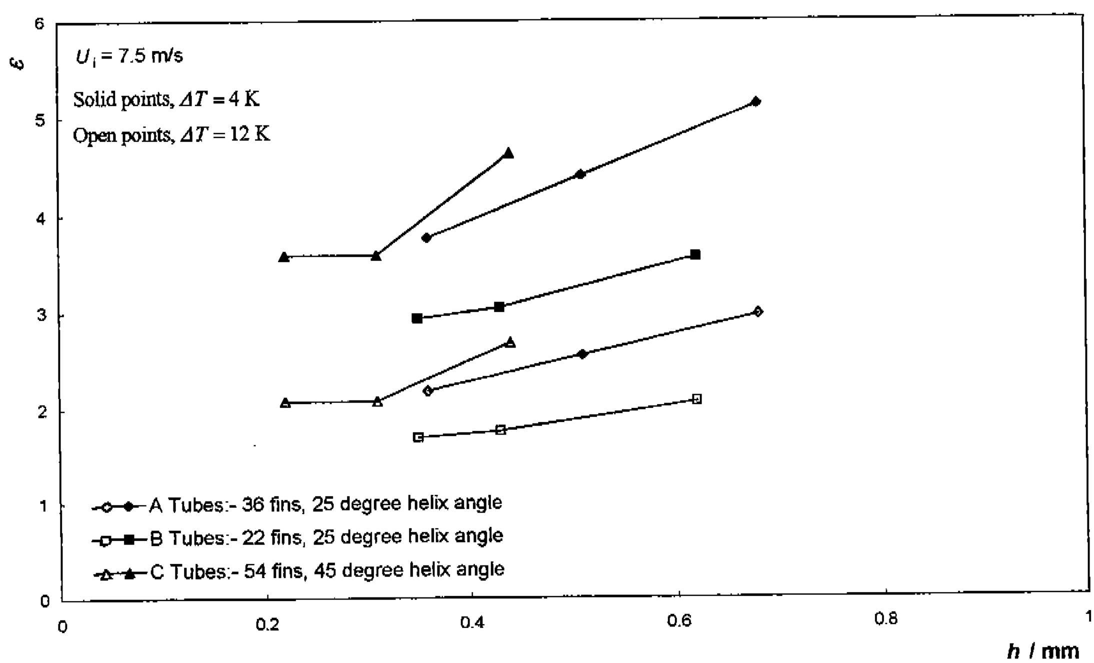

Figure 4 - Variation of Enhancement Ratio with Fin Height (note points are joined by lines to guide the eye)